N74-10297

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Paper No. 67

A NOVEL FACILITY FOR RADIANT HEAT TRANSFER EXPERIMENTS

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ABSTRACT

In order to perform a special experiment in radiant heat transfer a unique facility was designed and built. The fundamental design parameters included the capability for achieving rapid initial temperature rise and for establishing effective maximum temperature as high as possible for continuous operation. In this presentation, the design procedures and operation of the facility are discussed. A particular experiment measuring the emittance of a titanium alloy is summarized to exemplify one application of the facility and finally, the versatility of the equipment is pointed out in discussing other potential areas of applicability.

INTRODUCTION

The particular requirements of a special radiant heat transfer experiment led to the design and utilization of a responsive and versatile test facility. The unit evolved is shown in Figure 1. Based on the requirements of the experiment, an initial rate of rise in temperature of 700° per second and a nominal maximum temperature of 1700°R were selected as design criteria for the radiant source. The rapid rise in temperature was obtained by applying the combustive heat provided by an oxy-acetylene flame to a metallic radiator with minimal heat capacity. The radiator, a thin walled metallic cylinder, was mounted on a rotating platform in order to distribute the heating as uniformly as possible. The acetylene flame issued from 34 individual jets spaced along each of four columnar manifolds, washing the length of the cylinder. The design permitted the columnar manifolds to swivel, controlling the absolute level of the temperature at which the cylinder equilibrated.

Among the experiments for which the facility was employed was one in which the total emittances of various metal radiators in cylindrical form were determined. The temperature of each metal cylinder was recorded by two thermocouples affixed to its surface and the thermal flux density emitted was measured by a calorimeter placed close to the interior wall of the cylinder.

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One cylinder was constructed from a titanium alloy. The characteristic emittance varied from 0.4 to 0.6, indicating some sensitivity to temperature, and suggesting a slight progressive variation at an approximately steady temperature.

In addition to the experiment involving the imposition of a controlled temperature profile on the radiative source, quite a number of other experiments can be performed in the oxy-acetylene facility. One of these, the measurement of total emittance, is described in this presentation. As with the several other studies performed in this facility, the experiment yielded useful information. Additional applications for which the facility might be used are reviewed.

DESIGN CRITERIA

The radiant facility was created to simulate with reasonable fidelity, variable temperature profiles. The principal restriction on the profiles was the establishment of moderately high temperature levels in a brief period of time. A design objective was established which sought to achieve an initial rate of rise in temperature of 700°F/sec. The nominal value of the maximum operating temperature of the facility was identified as a second design criterion. While the highest possible value was desirable, the design value had to consider the high temperature structural capabilities of the radiating cylinder. A conservative value of 1700°R was selected. The two elementary criteria are shown in Figure 2.

Having established design objectives, the actual design procedures involved first an estimation of the heat transfer mechanisms involved, to ascertain the quantity of heat required to achieve them. Thereafter, the combustion of acetylene was reviewed to determine that the process would indeed provide the necessary heat at adequate rates. When the technique appeared to satisfy the criteria the quantities of oxygen and acetylene required were used as the bases upon which the physical design was developed. The design provided a system for distributing the combusting acetylene and incorporated a system of target rotation to approximate uniformity of heating on the surface of target cylinder. In addition to satisfying requirements of the original experiment, mechanical design attempted to maximize versatility in the operation of the facility.

THERMAL DESIGN

To determine, ultimately, the quantities of oxygen and acetylene required to achieve the projected performance of the facility, the demands of the various mechanisms consuming the thermal energy generated were examined. These include

- Heat capacity of the cylindrical radiator
- Natural convection
- External radiation
- Internal radiant exchange

It is obvious that the relative contributions of the various factors will be time dependent. The net effect of all the factors taken as a function of time constitute the thermal demand of the system, which must be satisfied by the combustion of the acetylene.

The first step in treating with the contributing thermal process was to prepare design curves of the individual processes. Since the heat capacity of the cylindrical radiator appeared to be the most significant mechanism initially, it was examined first. The cylinder was to be made of metal. A titanium alloy was one material selected. Preliminary sizing requirements fixed the cylinder at twelve inches in diameter and three feet in height. To minimize the thermal capacity, a wall thickness of 10 mils was assumed. Using the appropriate properties from Table 1, the effective heat transfer rates necessary to sustain specified temperature rise rates were computed using the calorimetric equation.

$$\dot{q} = \rho \, C_p s \, \frac{\Delta T}{\Delta t} \tag{1}$$

The design graph (Figure 3) indicates the quantities of heat necessary to produce specified increases in temperature per unit time. The data reflect the sensitivity of the relationship to variable specific heat.

Another mechanism which utilizes the input energy is thermal radiation. It was assumed that the combusting acetylene contributed a negligibly small thermal radiation input into the cylinder. Most of the heat transfer entered through convection. However, as higher temperatures were attained by the metal cylinder, increasingly large amounts of thermal energy were lost through reradiation. Using the Stefan Boltzmann equation, the energy reradiated per square foot of surface area can be calculated parametrically. The equation.

$$\dot{q} = 5.0\epsilon \left[\left(\frac{T_w}{1800} \right)^4 - \left(\frac{T_a}{1800} \right)^4 \right] \tag{2}$$

was evaluated for emissivities of 1.0, 0.8, and 0.6. The numerical results are illustrated as design curves in Figure 4.

Along with the foregoing modes of heat transfer, a convective loss in the system was expected. Although the metal cylinder was designed to rotate the speed is minimal. Therefore, to a first approximation, the heat transfer was treated as free convection. The system was represented as a vertical wall. According to Fishenden and Saunders (Reference 1), the heat transfer coefficient correlates as follows:

$$h' = 0.28 \, (\theta / l)^{1/4} \tag{3}$$

for laminar flow and

$$h' = 0.30\,(\theta)^{1/4} \qquad (4)$$

for turbulent flow. Turbulence is defined to exist for flows in which the product of the Grashof Number and Prandtl number exceeds 10^8, numerically. The rate of heat transfer is then given by:

$$\dot{q} = \frac{h'\,\theta}{3600} \qquad (5)$$

The Grashof Number was obtained at an average film temperature

$$T_f = \frac{T_w - T_a}{2} \qquad (6)$$

and is given by

$$N_{gr} = \frac{g\,\beta\,l^3\,\theta}{\upsilon^2} \qquad (7)$$

Taking advantage of the fact that for a gas, β is proportional to $1/T$, the product of Grashof and Prandtl numbers were calculated for a range of representative average film temperatures. It was determined that for the conditions of operation of the cylinder, the flow was, according to the above criterion, predominantly turbulent (i.e., $N_{gr}\,N_{pr} > 10^8$). Consequently, the film coefficient and the consequent rates of heat transfer were based on equation (4). The values of the rate of heat transfer for a representative range of temperatures are presented in the form of a working design curve. (Figure 5).

Having compiled data assessing the potential contributions of the various modes of heat transfer, it is appropriate to estimate the contributions of each and establish a time dependent profile of heat transfer. At the initial application of heat (time zero), the titanium is at ambient temperature. Hence, both radiant and convective heat transfer will be zero. Therefore, initially all of the heat transferred will be consumed in satisfying the heat capacity of the titanium. To assess the subsequent behavior, it is convenient to define a coefficient of heat transfer at time zero. A family of coefficients for arbitrary temperature differences is shown in Figure 6. The design point has been indicated. It may be noted that the design rate of heat transfer, 40 Btu/sec ft^2, exceeds the 25.2 Btu/sec ft^2 that is demanded by the titanium as depicted in Figure 3. It is inherently impossible for the transfer of heat to proceed with an efficiency of 100%, a condition implicit in the evaluation of the heat capacity requirements of titanium. Therefore, the rate of heat transfer selected as an objective exceeded the ideal, implying an assumed efficiency of approximately 60%. Since, as is summarized in Table 2, typical flame temperatures of combusting acetylene exceed 4500°R, at time zero, the initial difference in temperature between the combusting gas and that of the cylinder metal will be at least 4000°R. Assuming this difference, the coefficient of heat transfer to the cold wall used for initial design was 0.01 Btu/sec ft^2 °F. This value is based on the elementary definition of the heat transfer coefficient. In furnace analysis, the rate of heat transfer between combustion gases and the charge in a furnace is represented by a formula of the form:

$$\dot{q} = h (\Delta T)^y \tag{8}$$

Empiric treatments have provided for variations in the performance of the combusting environment by introducing to Newton's expression a variable exponent for the temperature difference. However, according to Thring, (Reference 2), the exponent is in most cases unity, and in all cases, near to unity. Hence, the simpler assumption is consistent with furnace practice.

The combined heat transfer was assessed by determining first the changes in temperature of the metal of the cylinder as a function of time, due to heat storage. This was treated incrementally using arbitrary, small temperature rises. The new driving difference in temperature was then applied to each subsequent calculation. Each incremental temperature rise was calculated from that preceeding it, using the equations

$$t_n = \frac{(\rho C_p s)_n (T_{n+1} - T_n)}{h (T_s - T_n)} \tag{9}$$

which evaluates an incremental time associated with the assigned temperature rise, $(T_{n+1} - T_n)$, of the cylinder, and

$$\dot{q}_{n+1} = h(T_s - T_{n+1}) \frac{T_m - T_{n+1}}{T_m - T_n} \tag{10}$$

which provides the data necessary to continue with the next incremental evaluation. The procedure iterates using each incremental value. The numerical values obtained are presented in Table 3. The table indicates a theoretical time rate of rise of 1190°/sec can be expected initially, although the rate declines very rapidly.

The time profile defined by the foregoing analysis was employed to approximate the thermal demand required by the other modes of heat transfer. Utilizing the appropriate design curves, values for both radiant and convective heat transfer rates were obtained for the incremental times previously calculated. The values are given in Table 4. Radiant exchange, assuming an emissivity of 0.6 was charged to both faces, although the subsequent test programs were performed under different conditions. Convective heat transfer rates were increased three fold to compensate for possible enhancement of heat transfer due to the rotation of the cylinder. A compilation of the data from Table 4 is presented graphically in Figure 7. The information must be interpreted for application to the operating facility. An incident heat transfer rate of 40 BTU/sec ft^2, were it to be achieved, would be more than adequate to satisfy the first criterion. However, the primary purpose of the design based on 40 BTU/sec ft^2, as previously stated, was to protect against unanticipated losses and the inefficiency of the system.

A natural consequence of establishing a specific design rate of heat transfer is to determine the quantities of fuel and oxygen which will be required to maintain such a rate. Commercial applications, of which the facility design made use, employ the principals, acetylene and oxygen, in a combustion process that departs from the stoichiometric reaction (Reference 3). Combustion is controlled to occur in two steps. Pure oxygen is supplied in equal volume with the acetylene. The initial oxidation process involves these volumes.

$$C_2H_2 + O_2 \rightarrow 2CO + H_2$$

The products created are further oxidizable. These combine with oxygen from the air to produce the stoichiometric end products.

$$2CO + O_2 \rightarrow 2CO_2$$

$$2H_2 + O_2 \rightarrow 2H_2O$$

In this system, 40% of the required oxygen is supplied in pure form. The remaining 60% is supplied from the surrounding air. Thus only 15 volumes of air are needed to complete the combustion process. The two step process was chosen as most suited to the facility because it provides an additional measure of control over the thermal environment by adjusting the oxygen-acetylene mixture. A one-to-one mixture of oxygen and acetylene yields a "neutral" flame. An excess of acetylene yields the reducing environment, while an excess of oxygen will produce the oxidizing flame. In actual application it proved convenient to use the neutral flame mode of operation.

The one to one mixture of fuel and oxygen, together with the supplementary air produce the operating flame. Based on visual observation of an oxy-acetylene flame it was estimated that a single jet, although dispersed over a larger area, provided the nominal heat transfer rate over an area one inch in extent. Consequently, for a 36" multijet column, there were 34 jets located on one inch centers. Four columns were specified. For the rotating cylinder, each column contributed a proportionate share of heating. The total surface area of the cylinder was 9.42 ft^2 and the maximum heat transfer conservatively estimated to satisfy the design criteria was 40 Btu/sec ft^2. Since the heating value of acetylene is 20,000 Btu/lb, (See Table 2) the mass flow needed to satisfy the rate of heat transfer is calculated from

$$\dot{q} = \frac{h_v (\dot{w})}{A} \tag{11}$$

Upon rearranging,

$$\dot{w} = \frac{40\ (9.42)}{20 \times 10^3}$$

$$\dot{w} = 18.84 \times 10^{-3} \quad \text{lb/sec}$$

A like amount of oxygen is used in the combustion as conventionally provided; therefore, the total mass flow will be 37.7×10^{-3} lb/sec. Since there are 136 orifices, from each will issue 27.7×10^{-5} lb/sec of mixed gases. There remains the problem of sizing the orifices. Since the gas mixture is variant from gases commonly tabulated, the properties have been separately assessed. Since commercial mixing ratios were to be used in the facility nozzle calculations were based on equal volumes of acetylene and oxygen being emitted from the nozzles. Using the individual gas properties, the properties of a mixture were assigned according to the data in Table 5. The data were given in Reference 4. Note that the gas constant R differs from that employed in the nozzle calculations (R_u) by the gravity acceleration constant. The properties are representative of gases at 540^{o}R. These values were used in the subsequent flow calculations. To better define flow rate, a choked flow was assumed. Each orifice was designed to provide fixed mass flow rates above the critical pressure. The flow rate is calculated with the equation (Reference 5).

$$\dot{w} = \frac{144\ \rho o}{R_u\ T_a} \left(1 + \frac{\gamma - 1}{2}\right)^{-1/(\gamma - 1)} \sqrt{\frac{2\gamma R_u T_a}{\gamma + 1}}\ gA \tag{12}$$

Using the properties of the gas mixture, the mass flow per unit area was obtained allowing for the orifice efficiency of flow. The nozzle size of each of the 136 of the jets was determined to be 0.005 inches. Since this was a value impractical to realize because of manufacturing costs, an arbitrary diameter of 0.020 was specified for manufacture. It was realized that the increase in diameter would result in a significant increase in gas consumption and would very likely degrade facility performance.

MECHANICAL DESIGN

The mechanical design of the radiant heat transfer facility had two objectives. The first was to provide a system which satisfied the design criteria and adequately met the requirements of the original test program. The second objective went further. It attempted to incorporate features which made the system more adaptable and more versatile. Adaptability was required to resolve on-site developmental problems and versatility was desired to make the facility useful for investigations beyond the initial one.

The basic structure and salient features of the facility are illustrated by the cutaway view (Figure 8). The outer flanged housing supports and protects the drive motor. The flanged column inside is the base for a fixed platform on which target specimens are mounted. Supported on the column through roller bearings is a large flat plate topped with insulation which rotates freely, driven through a system of gears by the electric motor. The radiating cylinder mounts atop the rotating top plate. The outer edge of the plate is supported on rollers mounted atop the flanged housing. A large diameter slip ring assembly was mounted to the underside of the driven gear and permitted thermocouple data to be transmitted from the rotating metal cylinder to stationary brushes. Model instrumentation lines were brought down the interior of the central column. Four supportive brackets with bushings each received a shaft onto which mounted a horizontal bracket structure. A clevis joined the columnar torch heat to the end of this bracket, and a turnbuckle assembly restrained the two parts in relative positions one to another. Conventional oxy-acetylene torch equipment was purchased to supply the facility torch heads. The normal torch head was removed and the handle, with mixing equipment and valving retained, was brazed to each columnar torch head. The 34 jets were screw fitted into the column and sealed with silicone O-rings. A band of copper tubing looped around each header to provide water cooling.

Although not shown, each shaft passing through the supportive brackets on the periphery of the flanged housing was equipped with a sprocket, and all the sprockets were joined by a sprocket chain. Thus rotation of one torch head resulted in the united rotation of all. Movements were produced by manual control.

Sizing the facility was dictated principally by the requirements of the original test program. The metal cylinder serving as radiator was sized to accomodate the test specimen and the number of jets in the columns were fixed as soon as their interval was established. As already noted, flow rates and thermal parameters consistent with the physical geometry of the housing were determined. The thermal calculations merely indicated that for the size proposed, the energy requirements apparently were attainable.

The features providing flexibility were numerous. Several are enumerated to emphasize their advantages.

1. The gear system was designed to accept any of seven pairs of gears to provide variations in speed of cylinder rotation. Although only one was used, they offer opportunities to investigate in detail the effect of varied convection flow characteristics.

2. Orifices were threaded individually into the torch heads. Modification of orifice sizes was possible with little difficulty. Obviously, there are limits in size which could not be exceeded, however.

3. The use of turnbuckles proved very helpful. They not only restrained the columnar torch heads in their proper position but they also provide a relative find control on changes in the alignment of them.

4. The ability to align the columnar torch heads on either side of the vertical anticipated one problem which required alleviation. When in the vertical position, the jets tended to produce a temperature gradient on the cylinder from top to bottom. The cylinder ran hotter at the top. By retracting the top of each column so that it was inclined outward at an angle of six degrees, a remarkably gradient - free vertical profile was observed on the cylinder.

5. The radiating cylinder was very simple to replace. Any formable sheet metal could be rolled and spot welded to produce a replacement cylinder. In some cases (aluminum for example) special consideration of the maximum operating temperature must be given.

6. The water cooling lines affixed to the torch head columns were installed to reduce the possibility of preignition of the oxygen-acetylene mixture in the headers. They were apparently successful; no incident of flash back was experienced.

The facility was furnished oxygen and acetylene from a bank of bottles organized in independent groups to supply each of four sets of inlet hoses leading to the mixing manifolds brazed to the columns. The gas system manifolded three bottles of acetylene and two of oxygen to supply each column (Figure 9). Individual torch columns could be ignited, adjusted, and even calibrated independently.

PERFORMANCE

The initial test for the complete system involved varying the temperature after achieving the nominal maximum value. The variation was obtained by manually swiveling the torch headers to vary impingement of the combustion gases. Guidance for such manipulation was provided by the responses of an IRCON two color pyrometer. In order to make more determinate the temperatures reported by the pyrometer, the outer surface of the cylinder was blackened. Acetylene soot was deposited by applying the torches. Obviously, a black radiating surface on the outside lowered the effective maximum temperature attainable, but permitted more meaningful control of transient temperatures. Nevertheless, the detailed data obtained during the initial period of heating (Figure 10) indicate the mean heating rate exceeded the design value over the first two seconds. Indeed, the experimental data are not only self consistent, but demonstrate a relatively close approach to design was achieved. The complete test profile is shown in Figure 11. Upon observing the temperature to reach 1700^{o} initially, the torch heads were swiveled to an off-center position resulting in the recorded decline. The variations of the cylinder temperature profile are indicative of the sensitivity of the cylinder in response to changes in the input heating.

EMITTANCE EXPERIMENT

The facility may be employed for a variety of experiments. One which has been performed investigated the emittance characteristic of a titanium alloy. In the original experiment involving the titanium alloy radiating cylinder, an emittance of 0.6 reported in the literature was employed. During subsequent tests of the emittances of various materials, it was convenient to make a measurement endeavoring to verify independently the value reported. The radiating cylinder itself became the test specimen. However, unlike the original cylinder, the ones used in this study were only six inches high. In order to minimize possible convection effects a close fitting cover was installed atop the fixed platform of the facility which effectively closed the upper end of the cylinder. Schematically, the system is shown in Figure 12. The specimen was inserted in a recessed base ring to raise it and more fully utilize the surface of the cylinder. Two thermocouples were welded into the inside of the test cylinder to monitor the specimen temperature. Signals were transmitted through the slip rings installed on the facility to the recording instrument.

Inside the test cylinder a second heavy walled cylinder of steel supported a commercially available thermal flux detecting calorimeter in a hole midway down the length of the cylinder. The gauge, calibrated with a radiant source, was furnished with a coating of 0.825 emittance, and had a capability of up to 10 Btu/sec ft^2. Basically, it was a Gardon gauge, which functions by generating a calibrated electrical output resulting from the temperature gradient between a constantan foil and a copper heat sink as shown in Figure 13. In the instrument employed, the heat sink is kept at constant temperature by water cooling.

During the test, the acquired data consisted of the temperature of the titanium cylinder and the flux absorbed on the face of the Gardon gauge. The data obtained from the two thermocouples on the cylinder was averaged. The thermal flux to which the Gardon gauge responded was based on the calibration of the electrical output provided by the manufacturer. The two sets of data are coplotted in Figure 14 as a function of time. The test duration was 125 seconds in an effort to establish complete thermal equlibrium.

The rotating cylinder and the fixed one with the calorimeter mounted flush with its surface were less than ¼" apart in the region of the gauge. Although the two surfaces are close together it was estimated that of the convective heating developed along the surface of the rotating cylinder, very little was transferred to the massive heat sink and the water cooled gauge it contained. Hence, the entire measured response of the gauge was attributed to thermal radiation.

The experimental data were reduced by using the Stefan-Boltzmann equation rearranged to solve for an effective emittance

$$\epsilon = \frac{2.1 \times 10^{12} \dot{q}}{T_1^{\,4} - T_2^{\,4}} \tag{13}$$

The value obtained is, of course, comprised of two components: the geometry of the system and the actual emittances of the surfaces involved in the exchange. Two interpretations of the geometry were considered. The one case assumed the two surfaces to be parallel plates. The other treated the exchange as that between an infinite plane and a small element of area, representing the gauge. Both provide a geometric factor of essentially unity, according to Reference 6. Therefore, the primary effect will be that of the two emittances. Of the two, that of the calorimeter surface (ϵ = 0.825) is known. The other, that of the titanium alloy cylinder is unknown. The two are related as follows: for the parallel plates:

$$F_e = \frac{1}{1/\epsilon_1 + 1/\epsilon_2 - 1} \tag{14}$$

And for the plane and planar element:

$$F_e = \epsilon_1 \, \epsilon_2 \tag{15}$$

For arbitrary emittances on the detector the corresponding values of the cylinder were plotted for all values of the effective emittance measured. (Figure 15). As can be seen, the divergence between the two methods of treatment is not large, even at the mid-range of values. Using the experimental data and evaluating them in accord with the two models illustrated in Figure 15, two possible values of emittance were obtained both have been plotted along with the unadjusted effective emittance, in Figure 16. Taking the solutions to represent extreme in the evaluation of the cylinder emittance, the error is approximately $\pm 6\%$ over most of the time of the test. Note that as the experiment progressed, there was evidence of a gradual increase in the emittance. The final average value of 0.58 is consistent with the reported value employed in the design of the original experiment. The trend to increase in emissivity suggests that continued heating affects the radiation characteristics of the surface progressively.

POTENTIAL APPLICATIONS

The versatility of the oxy-acetylene facility, coupled with the simplicity of the equipment makes possible a variety of useful experiments. Consider the opportunities to use the variable flux capability. For any of a variety of metallic radiators, the response of complex target specimens can be studies. The "time constant" for the response of a multiple radiation barrier can be determined, as well as its net effectiveness in the steady state.

The variation of the emissive power of materials subject to change because of the thermal environment can be examined in both the transient and the steady states. Furthermore, the radiant behavior of composites consisting of substrate and overlay may be explored.

For very precise experiments, the exact contributions of the convective heat transfer mode must be determined, but accepting that as achievable, data may be acquired regarding radiant interchange factors for a variety of geometries, and additional high temperature emittance data can be obtained by the same general method as described for the titanium alloy. Obviously steady state values can be derived at each of a series of selected temperatures. Note that the titanium data were acquired at a temperature in excess of 2000°R. Obviously, the existing system has a capability to establish temperatures levels greater than the original design temperature.

CLOSURE

The oxy-acetylene radiant heat transfer facility was designed to perform a specific experiment. The design procedure and features have been discussed. As it developed, the concept which evolved resulted in a facility of great versatility. Various additional experiments were performed, one of which has been described briefly. That experiment used the facility to obtain a measure of the emittance of a titanium alloy at a specific level of temperature. As has been briefly summarized, other experiments in radiant heat transfer are possible. The capabilities of the facility far exceed those originally required and many additional experiments can be performed using the superior capabilities.

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TABLE 1 - PROPERTIES OF TITANIUM ALLOY (Ti - 6Al - 4V)

T °R	500	1000	1500	2000	2500
ρ , lb/ft^3			277		
C_p	.140	.167	.216	.283 e	.351 e
k	5.4	7.6	10.0	12.6 e	15.2 e
ϵ	0.6 (mean value)				

Values of C_p and k from Reference 4   e: extrapolated

TABLE 2 - THERMAL PROPERTIES OF ACETYLENE

Heating Value	1350 Btu/ft^3, 21449 Btu/lb	
	IN AIR	IN OXYGEN
Flame Temperatures	2325°C	2927°C
	2598°K	3200°K
	4676°R	5760°R
	4216°F	5300°F

Data From Reference 7

TABLE 3 - SUMMARY OF TRANSIENT HEAT TRANSFER, ALL MODES

	Time, Sec.	.101	.283	.448	.759	1.059	1.680	2.480	4.730
	Temp., ^{o}R	660	840	960	1140	1260	1440	1560	1740
	^{o}F	200	380	500	680	800	980	1100	1280
HEAT TRANSFER BTU/FT^2 SEC	$\dot{q}_{Cp}$	40.0	34.8	27.4	22.8	16.4	12.5	6.95	3.61
	$\dot{q}rr$	.07/.14	.15/.30	.22/.44	.40/.80	.62/1.24	1.20/2.40	1.70/3.40	2.55/5.10
	$\dot{q}conv$	.06/.18	.22/.66	.32/.96	.50/1.50	.62/1.86	.82/2.44	.96/2.88	1.18/3.54
	$\Sigma \dot{q}$	40.32	35.76	28.80	24.10	19.50	17.36	13.23	12.25

TABLE 4 - TRANSIENT HEAT TRANSFER FOR HEAT CAPACITY

n	T_n ^{o}R (T_{init})	$\dot{q}$ Btu/sec ft^2	T_{av} ^{o}R	Cp s	T/ t ^{o}R/sec	T_{final} ^{o}R	t sec	T ^{o}R (new)*	$\frac{T_m - T_{n+1}}{T_m - T_{n=1}}$
1	540	40.0	600	0.0336	1190	660	0.101	3880	0.895
2	660	34.8	750	0.0353	990	840	0.182	3700	0.750
3	840	27.4	900	0.0373	730	960	0.165	3580	0.637
4	960	22.8	1050	0.0391	580	1140	0.311	3400	0.482
5	1140	16.4	1200	0.0410	400	1260	0.300	3280	0.380
6	1260	12.5	1350	0.0428	290	1440	0.621	3100	0.224
7	1440	6.95	1500	0.0447	150	1560	0.800	2980	0.121
8	1560	3.61	1650	0.0465	80	1740	2.250	2800	-----

*Initial Value of $T = 4000^{o}R$

Tabular Solutions for Equations 8 and 9

TABLE 5 PROPERTIES OF COMBUSTION GASES

	C_2H_2	O_2	MIXTURE
MOL. Wt.	26.02	32	29.01
SPECIFIC HEAT	.409	.2193	.324
SPEC. HEAT RATIO	1.23	1.394	1.321
GAS CONSTANT, R	48.25	58.8	53.27

FROM REFERENCE 4

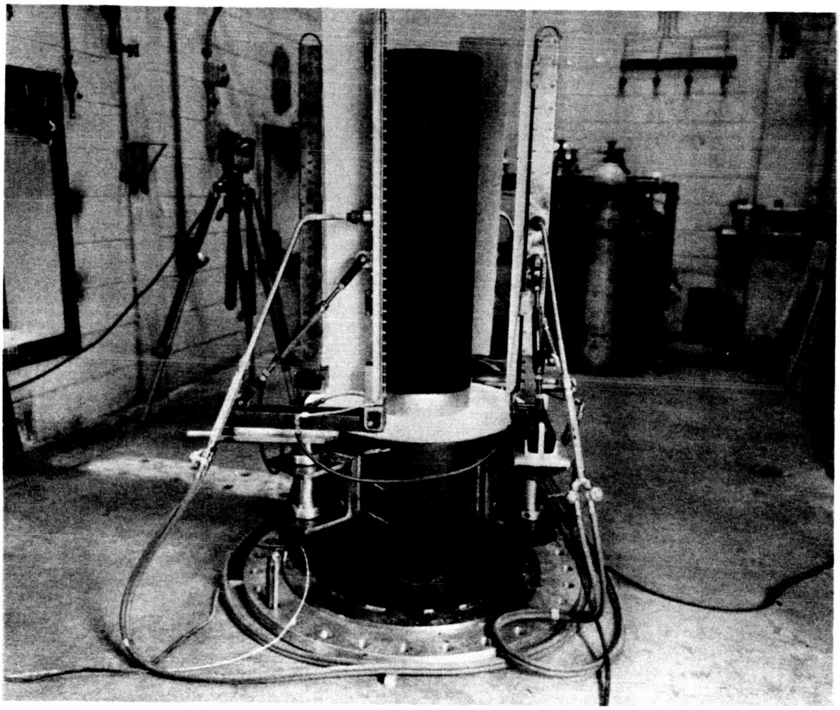

Fig. 1 General View of Oxy-Acetylene Facility

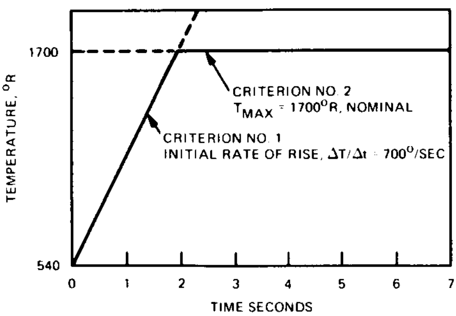

Fig. 2 Design Criteria

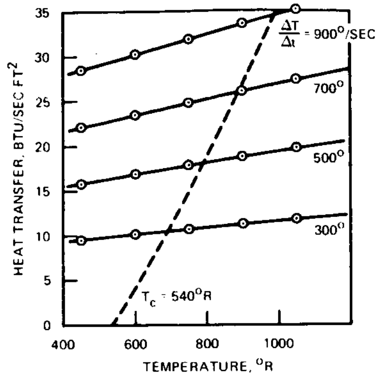

Fig. 3 Heat Transfer Rates for Given ΔT Rates

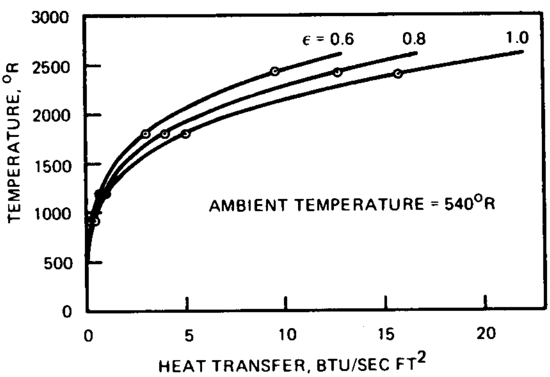

Fig. 4 Radiant Heat Transfer Losses

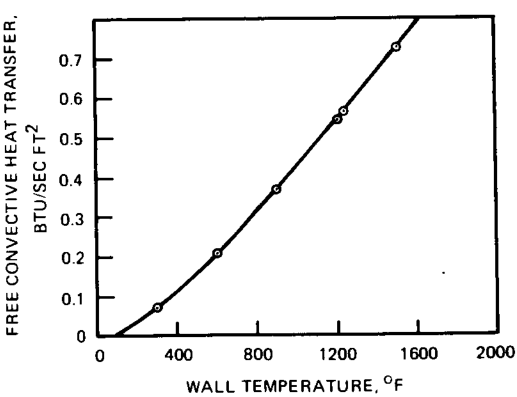

Fig. 5 Heat Transfer by Free Convection

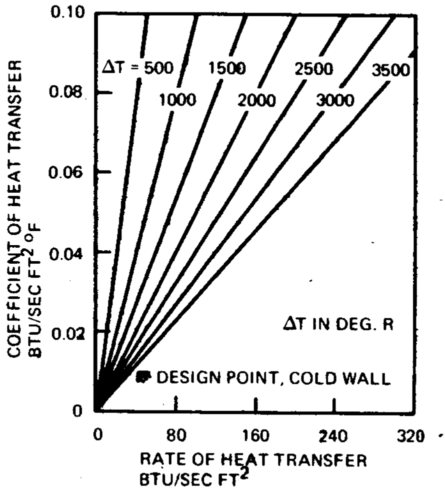

Fig. 6 Coefficients of Heat Transfer

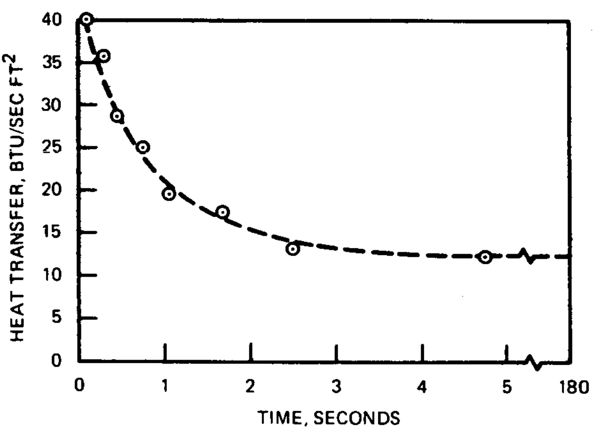

Fig. 7 Summary of Transient Heat Transfer

522-137 O - 73 - 55

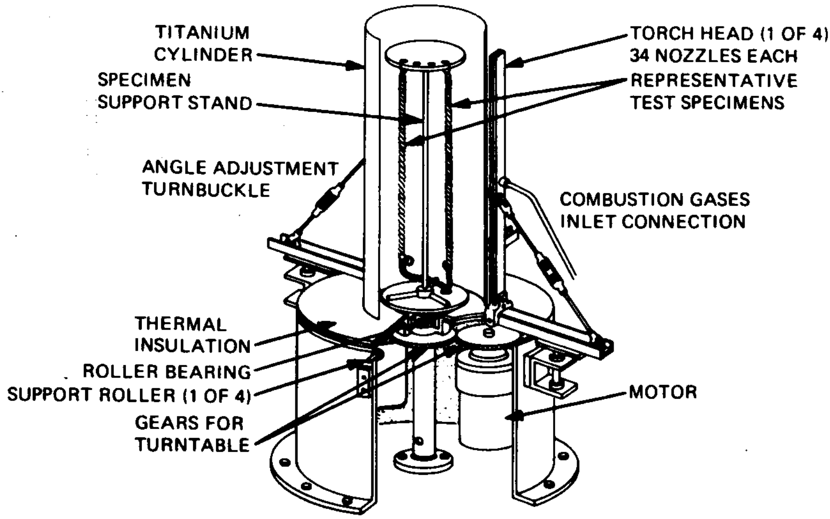

Fig. 8 Cut Away View of Radiant Test Facility

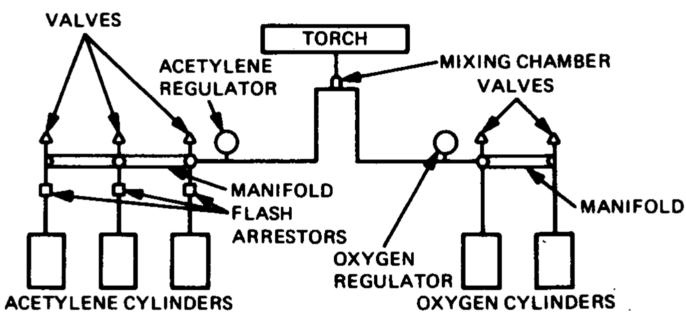

Fig. 9 Schematic of Gas Manifold System

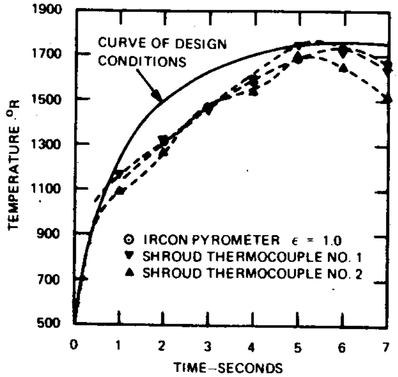

Fig. 10 Initial Temperature Profile

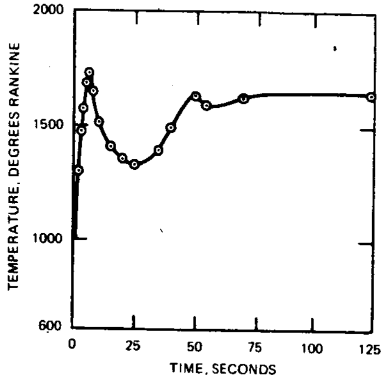

Fig. 11 Surface Temperature History of Cylinder

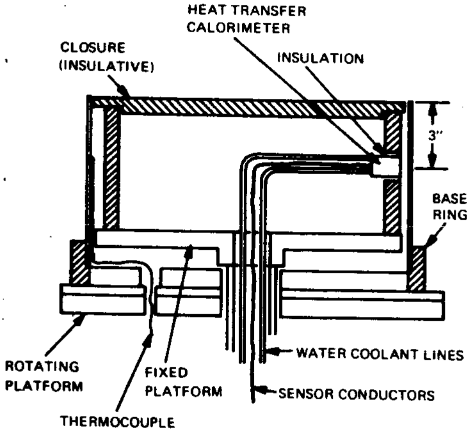

Fig. 12 Emittance Test Arrangement

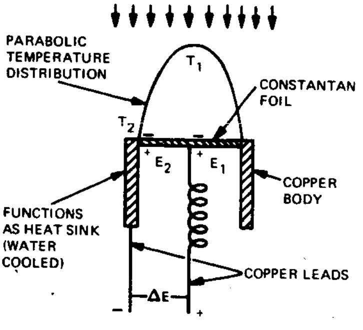

Fig. 13 Gardon Gauge Operation

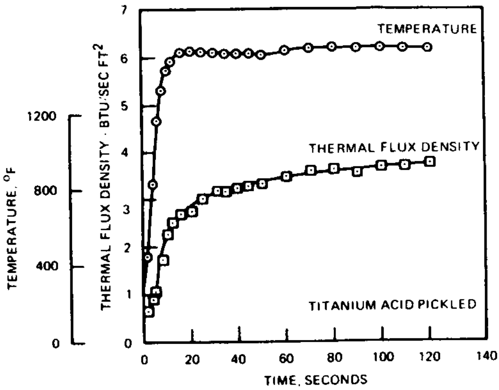

Fig. 14 Measured Response Data-Emittance Experiment

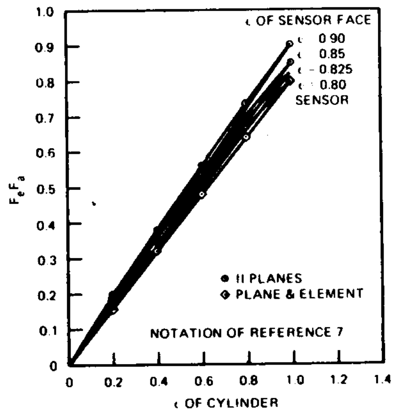

Fig. 15 Emittance Configuration Factors

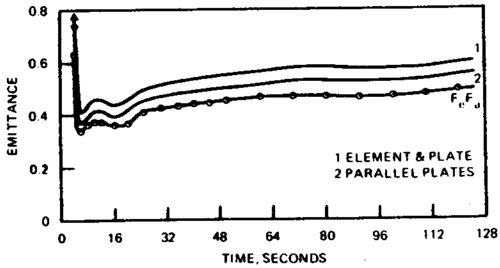

Fig. 16 Emittance-Titanium Alloy